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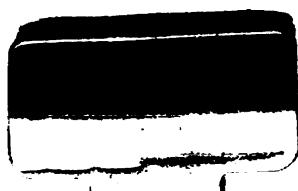
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B. E. ANDERSON
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AN INVESTIGATION OF THE EFFI-
CIENCIES OF A CRANBERRY
MARSH 16 INCH AUGER
PUMP

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AN INVESTIGATION
of the
E F F I C I E N C I E S
of a
CRANBERRY MARSH 16 INCH AUGER PUMP
under
Different Heads and at Varying Speeds

A Thesis submitted by

Earl Raimon Stivers
for the degree of
CIVIL ENGINEER

A Thesis submitted by

Bert Edwin Anderson
for the degree of
BACHELOR OF SCIENCE
Civil Engineering Course

A Thesis submitted by

Forest Eugene Wilterding
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Cranberry Marsh
16 inch Auger Pump.

CHAPTER 1.

INTRODUCTION.

The Cranberry Marsh 16 inch Auger Pump was designed by Mr. O. G. Malde, Superintendent of the State Experimental Station on the Cranberry Marsh near Grand Rapids, Wisconsin. The pump is purely a low-lift type of pump, designed to operate under heads not greater than six or eight feet. As it is used on the Cranberry Marsh it is driven by some type of gas engine, trial and experience determining the power required and the speed of operation. If it was found that the motive power at hand was not sufficient to lift the water the required height it was customary practice to lift the water in two or more stages.

In view of the fact that so little definite information concerning the performance of these pumps under varying conditions was available, it has been the purpose of this investigation to determine, if possible, the laws governing its efficient operation. The results sought for may be divided into three parts:-

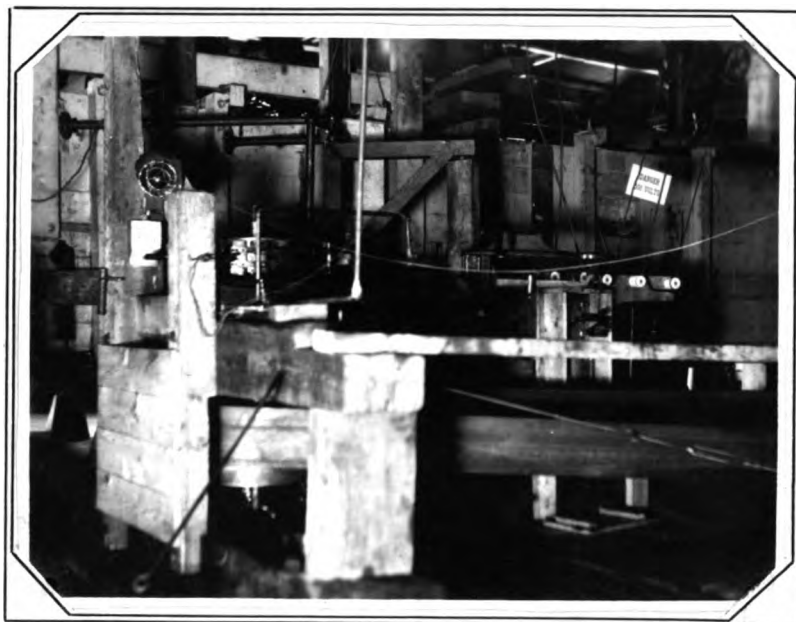
First: When the head to be pumped against, or the amount of water to be delivered, or both of them, is known; to determine the most efficient operating speed.

Second: To determine how closely the relations between head, speed, power, etc. for the centrifugal pump fit this type of pump.

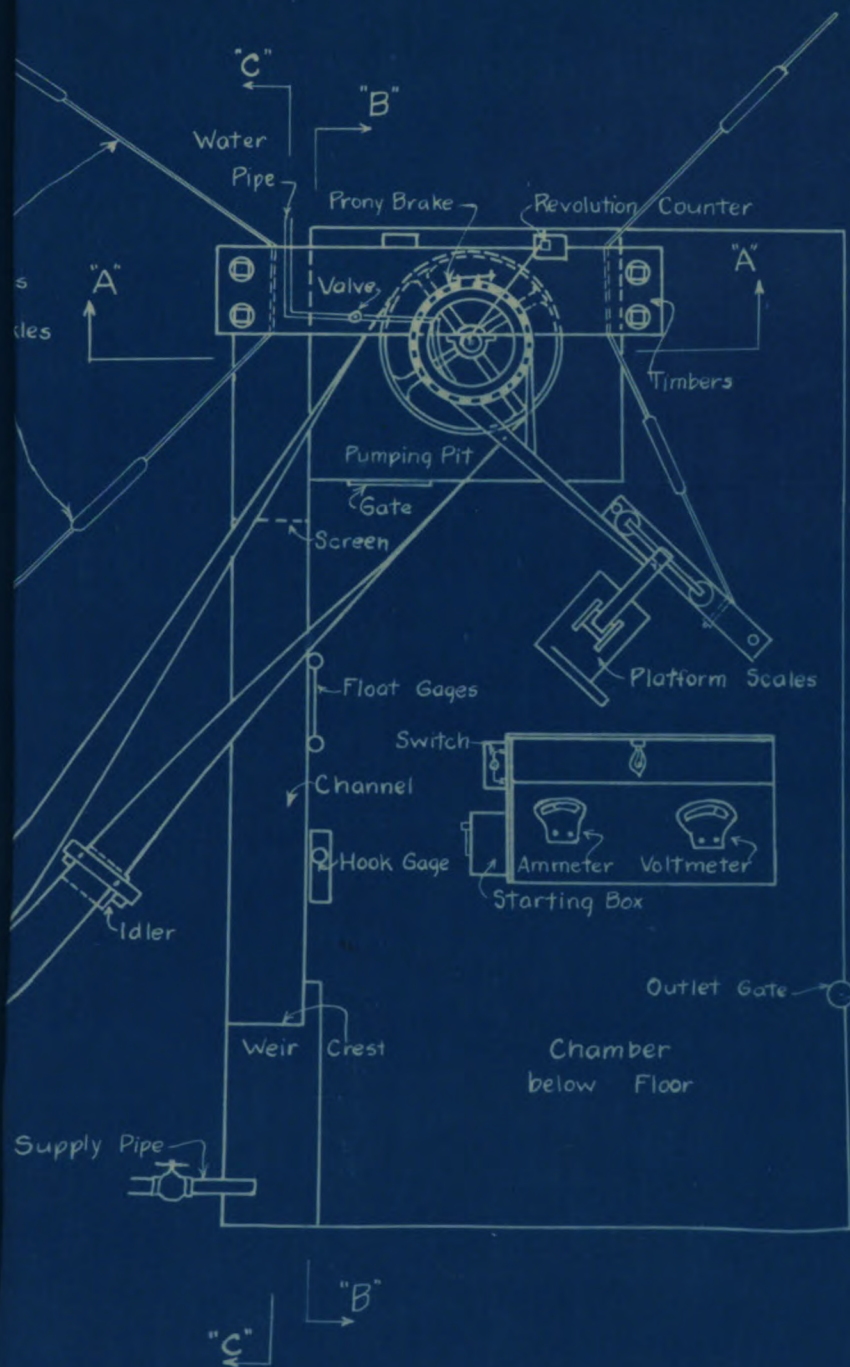
Third: To arrive at some conclusion as to what limits the use of the Auger Pump for practical pumping.

The authors are greatly indebted to Prof. C. I. Corp, for the many helpful criticisms and suggestions received. Thanks are also due to Mr. C. Deidrich, of the Hydraulic Laboratory, for his valuable assistance and suggestions in the construction of the apparatus.

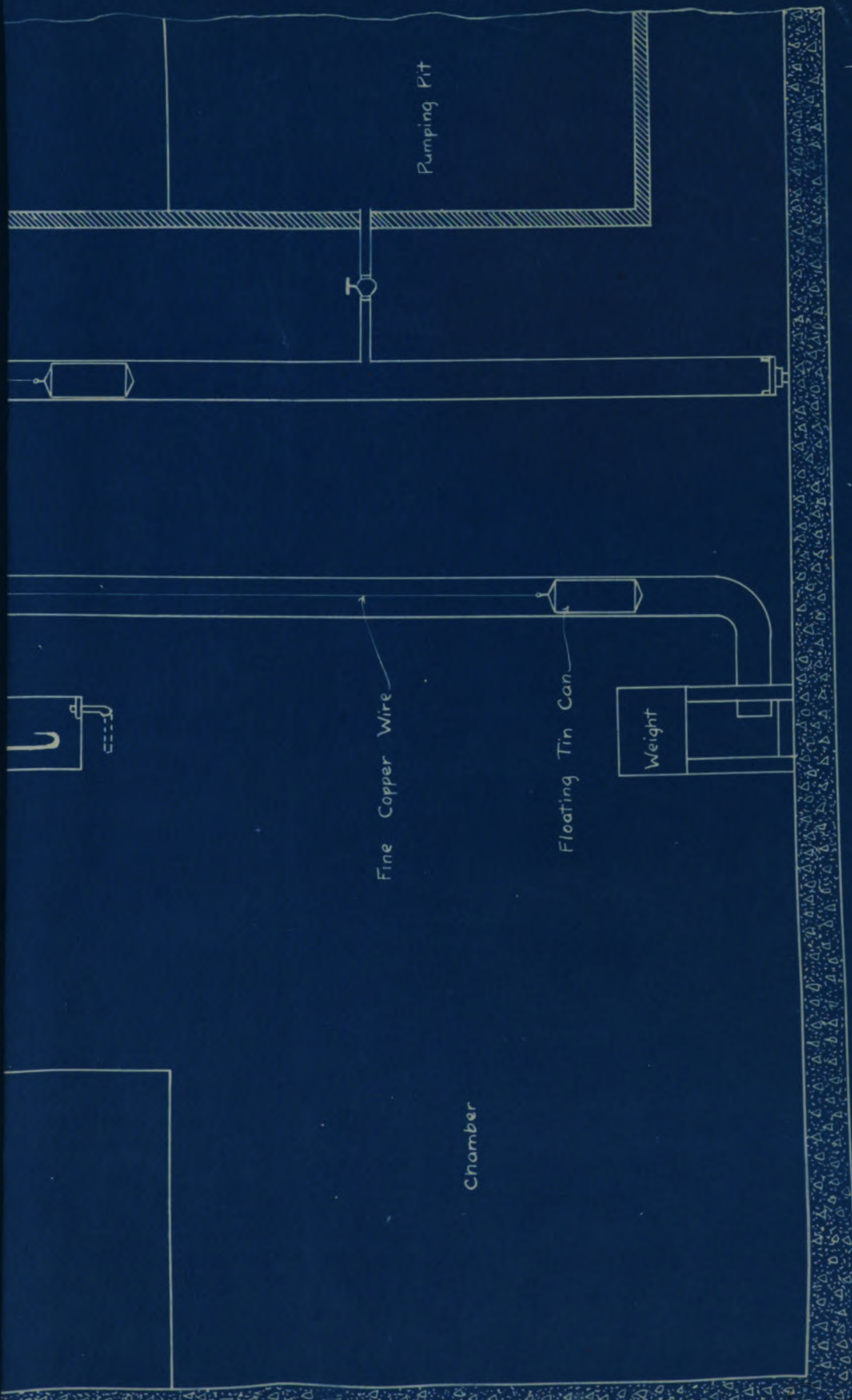
Plate A.

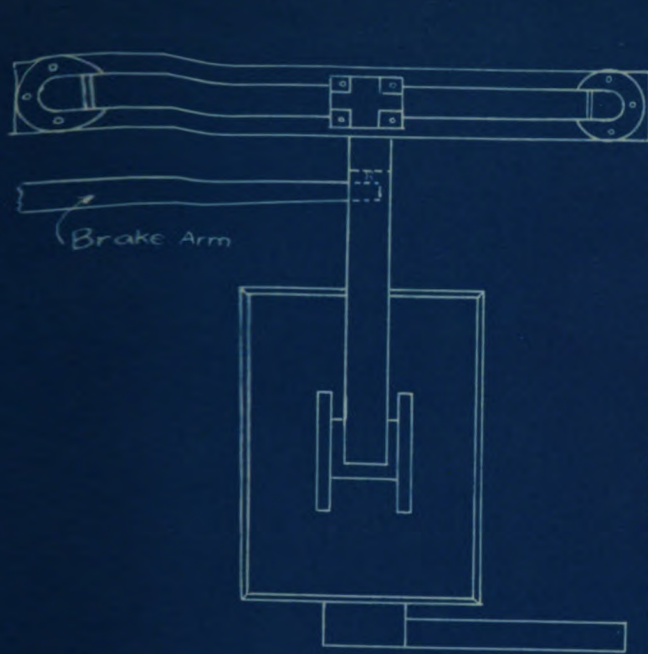


General View of Pump Test Apparatus.

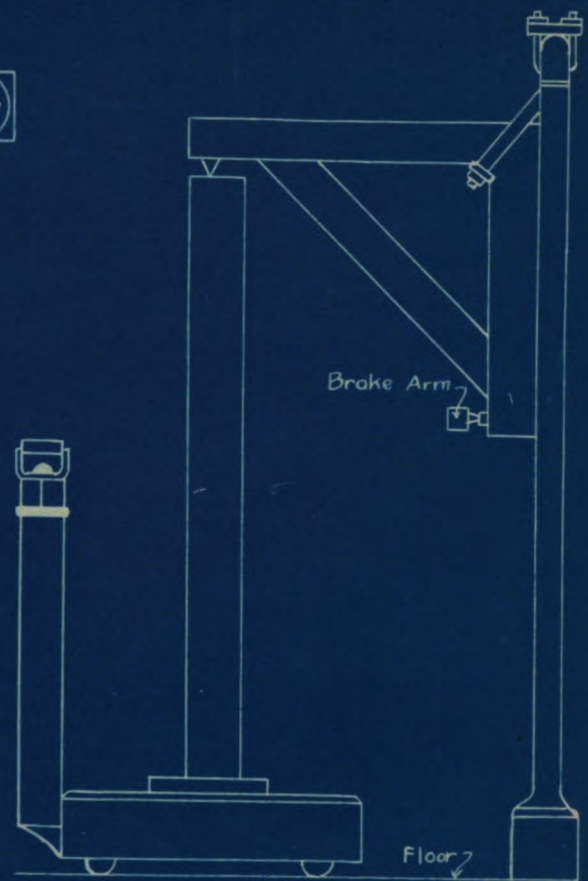


PLAN
FIG. 1.

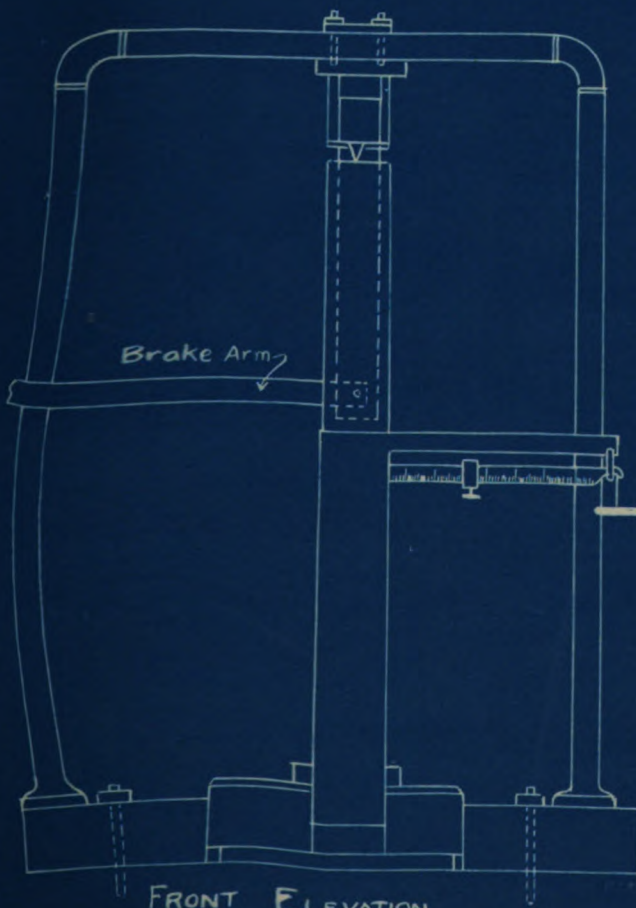




PLAN



SIDE ELEVATION



FRONT ELEVATION

DETAILS OF APPARATUS FOR WEIGHING BRAKE LOADS

FIG. 4.

CHAPTER 11.

A P P A R A T U S.

GENERAL LAYOUT - A general idea of the arrangement of the apparatus used in performing the tests can be had from Plate A. Fig. 1 is a plan and elevation of the apparatus, and Fig. 2 shows a cross section thru "A-A" of Fig. 1. Figs. 3 and 4 show the details of the float gages and the brake standard.

MOTIVE POWER - The motive power was furnished by a constant speed, direct current, shunt motor, rated at 550 volts, 25 amperes, and 1300 R.P.M. A view of this motor and the different sized wheels which were attached to its shaft in driving the pump is shown in Plate B. As shown in the plate, it was set on two firm runways, arranged so that in addition to the movement provided in the base of the motor itself, it could be moved in either direction to provide for the tension in the belt, when different sized pulleys were used to vary the speed at which the pump was operated.

VARIATION OF SPEED - In all, five pulleys were used; a 6 1/2 inch, a 9 inch, a 12 inch, a 14 inch, and a 15 inch, giving pump speeds of approximately 300 R.P.M., 390 R. P. M., 513 R.P.M., 600 R.P.M., and 640 R.P.M. A 28 inch steel pulley was used on the pump shaft.

Plate B.



Motor and different sized drive wheels.

BELT TRANSMISSION - As it was necessary to use a very long drive belt, and as it was found that a rubber composition belt best suited the damp conditions under which it was to be used, trouble was at first experienced in maintaining sufficient tension in the belt to keep it from dropping down and slipping off from the pulley on the pump shaft.

IDLER PULLEY - This difficulty was overcome by the use of an idler pulley supporting the slack side of the belt at about the mid point. The vibration of the idler formed small waves in the still basin causing some trouble in reading the hook gage, but with sufficient bracing this trouble and source of error was practically eliminated.

PUMP SHAFT - The pump shaft was fixed to two 12 inch by 12 inch horizontal timbers, which were clamped to the floor and braced by four stay rods provided with turn buckles. By means of these four stay rods with turn buckles a very fine adjustment of the pump shaft bearings was easily secured. The shaft was made up of two sections, joined by a shaft coupling. This was essential because of the necessity of removing the pump wheel when the brake runs were made.

PUMP WHEEL AND SHAFT SUPPORT - The lower end of the shaft was supported by a lignum vitae bearing, resting in a steel socket cap attached to the shaft. Just above this the pump wheel was attached to the shaft by means of three headless set screws. At the end of the

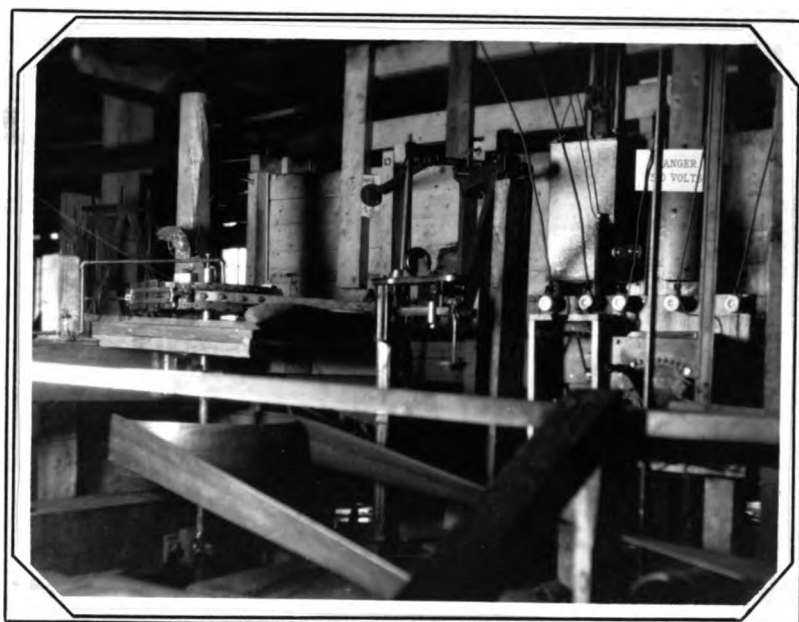
shaft and below the pump wheel a small circular bearing block was fastened by means of two ordinary set screws. The wheel turned in a cast iron casing, so that the joint was comparatively close. The floor of the pumping pit was 24 inches above the floor of the large water supply chamber.

BRAKE AND BRAKE WHEEL - At the top of the shaft and above the upper 12 inch by 12 inch horizontal timber a 20 inch brake wheel was fixed. To this a Prony brake with a 41 inch brake arm was attached when brake runs were made. The point at the end of the brake arm was made to press against the vertical leg of a bell crank that was swung by a knife edge from a standard. The horizontal arm of the bell crank transmitted to load to the scale platform through an upright post set on the platform. A view of this apparatus, used in determining the power input to the pump, is shown in Plate C. Considerable difficulty was at first experienced with the brake arm because it was made of a bar of wrought iron. This was due to excessive vibrations. One time the brake arm broke away from the bell crank, doing considerable damage. To prevent a recurrence of this and avoid the excessive vibrations a lighter brake arm was made of wood and the standard more securely braced.

A covering of tar paper was arranged on a framework just below the brake wheel, to protect the belt from the dripping water used to cool the brake wheel.

SPEED RECORDER - The speed was taken by a con-

Plate C.



Apparatus used in determining the power input
to the Pump Shaft.

tinuous revolution counter operated by an eccentric pin at the top of the shaft. Time was taken with a stop watch.

WEIR - The amount of water pumped was measured by a 22.4 inch weir, with suppressed end contractions. The weir was calibrated by checking it with a 2 foot weir that had been previously calibrated by Mr. C. P. Conrad and his party, as a part of their thesis. The methods and results of this weir calibration will be found in Appendix A. The head on the weir was measured by a standard hook gage fixed to a metal still-basin at the left of the weir-channel, a view of which is shown in Plate D.

The several methods of determining the zero of the hook gage may be found in Appendix B.

WEIR CHANNEL - To break up the currents, iron out the waves, and lessen as much as possible the turbulent condition of the water as it left the pump pit, baffles were set in the channel about five feet back from the weir and a floating raft was placed about two feet above the opening in the channel leading into the hook gage still basin. This is well shown in Fig. 3, Section on C-C.

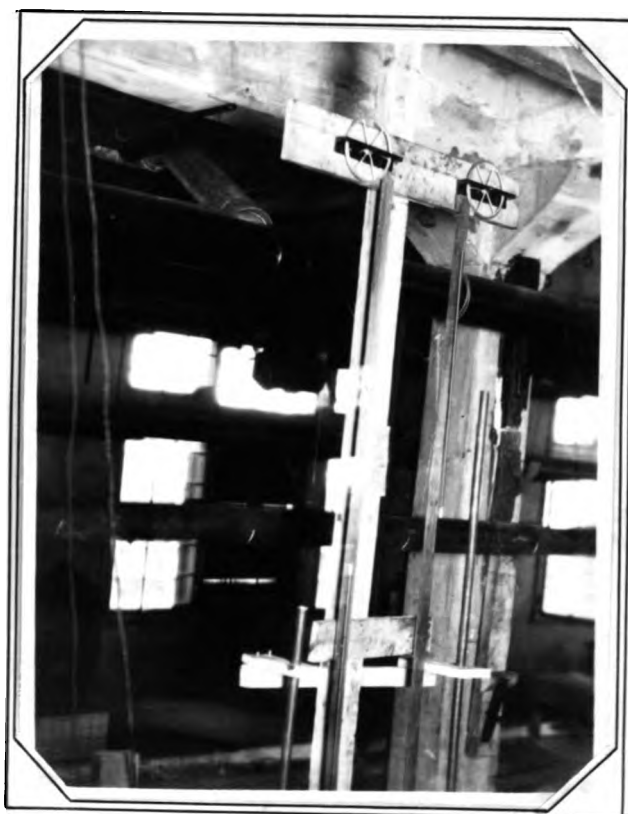
FLOAT GAGES- The difference in head pumped against, that is the difference in the height of the water in the chamber and in the pumping pit, was measured by means of float gages shown in Fig. 3, Section on B-B. These float gages, one connected directly to the chamber, and the other to the pumping pit, by means of pipe lines, were made of air tight metal cans, weighted at the bottom so they would float only in a vertical position. These

Plate D.



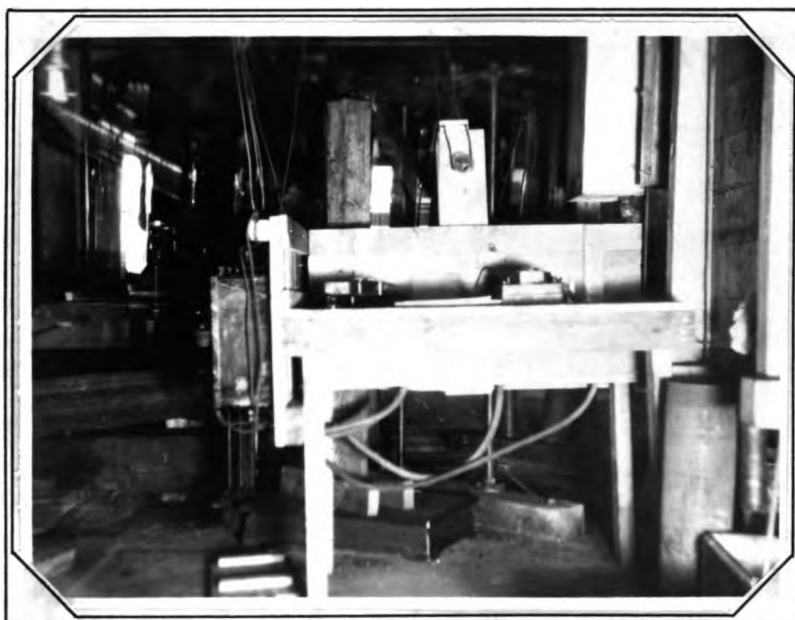
Standard Hook Gage.

Plate E.



Chamber and Pump Pit Gages.

Plate F.



Recorder's Table.

cans were counter balanced by $1/4$ inch round rods exactly two feet long and connected by a light copper wire running over two 5 inch frictionless aluminum pulleys, shown in Plate E. Steel graduated scales were placed in front of the rods so that gage readings could be read directly from either the top or bottom of the rod, notation being made in each case which was used. The floats were guided by two pipes, each $4\ 1/2$ inches in diameter. To protect the very light copper wire from being bumped into and broken it was found necessary to thread it through $3/4$ inch pipes extending up to within a few inches of the aluminum pulleys. The gage pipes were provided with valves which could be adjusted so as to prevent the floats from bobbing up and down too much due to the turbulent condition of the water.

The method of determining the zero of the float gages will be found in Appendix C.

POWER INPUT - The K.W. input of the motor was measured by an ammeter and a voltmeter arranged on the recorder's table, as shown in Plate F.

WATER SUPPLY - The chamber, preparatory to running a test, was filled by a 3 inch service pipe under 90 pounds head connected with the University Pumping Station, or by the low lift centrifugal pump in the Hydraulic Laboratory.

CHAPTER III.

METHODS OF TESTING.

TESTS - The tests were made in series, a complete test being made at a certain speed before the pulley on the motor was changed or the belt tension altered. Each test was numbered, as well as each run of every test, so that confusion was avoided when data was referred to.

BRAKE RUNS - The brake was attached to the brake wheel, after removing the pump wheel, and enough water was let into the pumping pit to just submerge the lignum vitae bearing and thus keep it cool. In making the brake runs one man was stationed at the table, and took readings of the voltage and amperage at intervals of about 10 or 15 seconds, and another man was stationed at the brake wheel where he could maintain a constant load on the scale by tightening or loosening the brake band. By means of a stop watch and continuous revolution counter observations were made for determining the R.P.M. It was found most convenient to call these to the meter recorder. For determining the speed the indicator readings were read from one to two minutes apart. Runs were made with net loads of from ten to sixty pounds on the scale. These were made at from five to ten pound intervals and at least three or four determinations of speed were made.

PRECAUTIONS TAKEN - Since in determining the efficiency of the pump, the ratio of the H.P. delivered to the pump shaft by the motor (obtained from the rating

curve) to the work done in pumping (height pumped against in feet times the weight of the water in pounds per second divided by 550) is used, small errors in performing the brake tests and in plotting the curves might have considerable effect on the final results. For this reason all possible care was exercised both in making the runs and in plotting the results. All computations for brake runs were made before any change in the apparatus was made. The reason for this was that if there was any doubt about any point on the rating curve check runs could be made under the same conditions until all the points lay on a smooth curve. Care was taken to keep the tension in the drive belt the same in both the brake runs and in the pump runs.

PUMP RUNS - After a brake run had been made with a certain speed the Prony brake was removed and the pump wheel attached to the end of the shaft and a series of runs made under different heads of water, starting with as low a head as possible and ending with as large a head as could be obtained with the apparatus and still have the bottom of the pumping pit entirely submerged. The lower limit was determined by the head at which back water interfered with the water dropping over the crest of the weir in the proper manner and under the same conditions in which it was calibrated.

ADJUSTMENT OF HEAD - The chamber was filled with water from the three inch supply pipe, or from the centrifugal pump in the laboratory, until the water in

the chamber and channel was at a level slightly above the top of the weir crest. The pump was then started, and, if after conditions became constant, the difference in height in the chamber and in the channel, by rough measurements, was about one foot, everything was ready for the first run. The required differences in head pumped against were adjusted by letting water out of the chamber through a sluice gate located at the bottom and in one corner of the chamber. Pump runs were made with heads varying from about one foot up to about 5.5 feet, which was the maximum obtainable with the apparatus used. The heads were varied in steps of one foot or less, so that enough points to make a good average curve were obtained.

CONSTANT PUMPING CONDITIONS NECESSARY - It was important that a run should not be started until the pumping conditions became constant. This could be determined by watching the hook gage, for when the head on the weir was almost constant the quantity of water pumped was constant. As the head pumped against was increased each run, the height of the water over the crest of the weir lowered each time and no readings were taken until it was certain that the height of the water over the weir would no longer decrease. Once or twice trouble was experienced in the results due to not having waited enough for constant conditions. As the floats sometimes stuck in the pipes special care was taken to see that the wires were perfectly free in their operation before any gage readings were

taken.

CONTINUOUS USE OF WATER - The water was drawn from the large chamber through the opening below the pumping pit, the bottom of which was 20 inches above the floor of the chamber, and lifted by the screw motion of the pump wheel, up through the pumping pit and into the weir channel. Here it was measured by means of the hook gage. From the weir the water dropped back into the supply chamber, where it was again pumped up over the weir.

PARTY AND DUTIES OF EACH MAN - Three men were required to make the pump runs. One man was stationed at the table and recorded the amperage and voltage, taking readings at intervals of about 15 seconds, and also reading the float gages at intervals of about one minute, each time being sure that the float gages were operating freely. As it was very important to obtain an accurate average of the amount of water pumped one man did nothing but read the hook gage, he trying to get a reading for the crest and trough of every other wave, the readings being taken about fifteen seconds apart. The third man was stationed at the revolution counter. Besides taking four or five speeds for each run, by means of a steel tape and plumb bob he obtained rough readings for differences of head pumped against, to be used merely as a check on the computations obtained from the readings taken by the float gages.

CHECK RUNS - All computation for each pump run

at a certain speed were made before there was any change in the apparatus so that check runs could be made if all the points did not lie on a smooth curve.

With the objects mentioned in the previous pages clearly in mind, namely:-

1. The determination of the most efficient operating speed;
2. The study of the laws of the small auger pump; and
3. The determination of the limits of efficient operation;

the authors, by various methods, were able to check up the results at various times, thereby ascertaining whether or not they were on the right track. It was not uncommon to go over many of the tests time and again when the results did not show what might be expected, in order to have absolute proof of any conclusions arrived at.

CHAPTER IV.

COMPUTATION OF RESULTS.

RATING CURVES OF MOTOR AND PUMP - As was mentioned before, all runs were computed and plotted before any changes were made in the apparatus. This not only lessened the chance for error but it also saved much time. The purpose of the brake runs was to determine the power delivered to the pump for a given power input to the motor. For the higher speeds the friction losses, etc., were practically constant so there is but little difference in the general shape of the curves. To assure accurate results brake runs were made at every speed. At the low speeds the friction losses are quite appreciable and any variation in the tension of the belt was certain to introduce a large error. For this reason special care was taken in making and checking the brake runs at the lower speeds.

The kilowatt input to the motor was taken as the product of the average values of the volts and amperes. The speed used was the average of four determinations at each load. The dead brake load was checked before each series of brake runs, and the net brake load was taken as the load indicated by the scale minus the tare load.

The brake horse power was determined by the

$$\text{formula:--} \quad \text{B.H.P.} = \frac{2 \pi L N W}{33,000}$$

where L = length of brake arm in feet ($\frac{41}{12}$).

N = R. P. M.

= . . .

-

=

-

W = net brake load on the scale in pounds

or B.H.P. = $.000651 N W$

The values of kilowatt input to the motor were plotted as abscissae and B.H.P. as ordinates. The curves in all cases were practically straight lines, as would be expected with the range of loads used.

PUMP DATA - In working up the results of the pump runs the average value of each part of the run was used. The kilowatt input to the motor was determined the same as in the brake runs. Knowing the kilowatt input to the motor the useful horsepower input to the pump was obtained from the curves plotted from the results of the brake runs.

ZERO OF HOOK GAGE - The zero of the hook gage, which was taken as 0.0398 feet, was the average value of a number of determinations, employing different methods as described in Appendix B.

ZERO OF FLOAT GAGES - The zero of the float gages, which was taken as 3.594 feet, was the average of many determinations, as described in Appendix C.

HEAD PUMPED AGAINST - The head in feet pumped against was determined from the float gage readings. After the zero of the float gages had been determined the constant difference between the tops of the two rods, when the water in the chamber and the channel are level, is known. This value, used in all computations, was found to be 43.594 feet. From an inspection of Fig. 3, Section

on B-B, it can be seen that as the head is increased, as the water in the chamber is lowered, the chamber float falls and the gage rod rises the same amount. As the pumpage falls off with increased heads the channel gage rod rises, although but very little compared with the change of the position of the chamber gage rod, due to the fact that a small change of head on the weir represents a large change in the discharge. The difference in head pumped against is found by adding to 3.594 the algebraic sum of pump and chamber gage readings, the former being considered negative and the latter positive.

H.P. DELIVERED BY PUMP - Knowing the head on the weir crest, the discharge in cubic feet per second was determined from the rating curve of the weir. (See Plate IX). The amount of water pumped in cubic feet per second, multiplied by 62.396 gives the quantity in pounds per second. The net horse power delivered by the pump is equal to the pounds per second pumped times the head in feet divided by 550.

EFFICIENCY - The efficiency of the pump is equal to the ratio of the H.P. input to the pump to the H.P. delivered by the pump. The percent efficiency is this ratio multiplied by 100.

SPEEDS - The speeds, or R.P.M., were not used directly in the pumping computations. These were taken merely for the purpose of finding the average speed of operation, which value was to be used in the final curves

and conclusions.

CHAPTER V.

DISCUSSION OF CURVES AND RESULTS.

The first object of this test was to determine the most efficient operating speed under different conditions.

The curves on Plate VI show the efficiencies of the 16 inch auger pump operated at various speeds, and at different heads pumped against. These curves were plotted from the pump run data, pages 37 to 40. It was assumed that the variation of efficiency could be shown by a smooth curve. When a point did not lie so that a smooth curve could be drawn thru it, it was considered that this was due to experimental errors, and a smooth curve approximating as closely as possible the true variation was drawn in.

An inspection of this group of curves will show the general shape of the efficiency curve for the auger pump. The curve for 397 R.P.M. gives the general shape of the whole curve. For the lower speeds, as 304 R.P.M. and 397 R.P.M. it was possible to obtain practically the entire range of efficiencies. With the apparatus used, at the higher speeds it was only possible to go slightly beyond the point where the maximum efficiency occurred, so these curves show only the first part of the entire efficiency curve.

There are two curves for the speed of 304 R.P.M. While they both have the same maximum efficiency, it

occurs at a slightly larger head in one case than in the other. This is due to the fact that the total power input was so small that the friction losses made up a very large part of it. As the two curves were obtained from two entirely independent runs, each including both brake and pump runs, the difference in the friction losses was undoubtedly due to the change in the belt tension.

As the speed of operation is increased the maximum efficiency becomes greater, and the head pumped against, at which the maximum efficiency occurs, also becomes greater. The maximum efficiency reached in these tests of the 16 inch auger pump is shown to be about 39% when operated at a speed of about 525 R.P.M., but the head pumped against is comparatively low, being about 3.75 feet.

With this variation in efficiency, due to changes in the speed, head, etc., table 1 was drawn up, all values being taken directly from the curves or from average results on the data sheets.

T A B L E 1.

Head in Ft.	Most Ef- ficient Speed R. P. M.	Discharge Cu.Ft.per Second	Discharge Gal. per Minute	% Efficiency to be expect- ed
<u>H</u>	<u>N</u>			
1.300	304	2.115	950	28
2.275	397	2.510	1130	34
3.750	523	3.980	1790	39
5.000	505	4.839	2170	38
5.740	653	5.000	2250	37

From this table the curves on Plate VII were plotted. It was at first thought that all these variables could be shown by a single curve having a graduated scale representing discharges and efficiencies. This was tried, but on investigation it was found that the scales varied so ununiformly that this was given up and the variation shown by means of several curves.

From these curves Table II was made for heads varying from one foot to six feet; showing the most efficient operating speed, discharge of water (gal. per minute), and the efficiencies to be expected.

T A B L E II.

Head in Feet	Most Effi- cient Speed R. P. M.	Discharge, Gal. per Minute	% Efficiency to be Expected.
1.0	275	920	26.5
1.5	325	975	29.0
2.0	370	1060	32.0
2.5	418	1225	35.0
3.0	460	1420	37.0
3.5	500	1670	38.5
4.0	540	1880	39.2
4.5	575	2040	39.0
5.0	605	2170	38.5
5.5	640	2230	37.5
6.0	665	2260	37.0

The curves of Plate VII indicate that the efficiencies drop off at higher speeds. The conclusion

reached in this report, however, is that while this may be true, speeds greater than 665 R.P.M. are not efficient speeds under a range of heads less than six feet.

LAWS OF THE AUGER PUMP - The second object of this test was to determine whether or not the laws of the centrifugal pump hold for the auger pump.

It is apparent that the height to which a centrifugal pump can lift water depends upon the speed at which it is operated. From the laws of the centrifugal pump we know that the peripheral speed of any runner may be expressed as

$$u = \sqrt{2gh} \quad \text{or}$$

$$\phi = \frac{u}{\sqrt{2gh}}$$

where u = peripheral speed in feet per second

ϕ = a constant

g = 32.2 ft. per second

h = head pumped against in feet

ϕ must have a most efficient value depending on the design of the centrifugal pump. It is then evident that the best speed for a given pump varies as the square root of the head. We can also see that for moderate changes in head the efficiencies of a given pump will vary but little as long as the speed is changed so as to keep the value of ϕ constant.

Then the law of the centrifugal pump may be expressed as

$$\frac{N}{N_1} = \sqrt{\frac{H}{H_1}}$$

where N = the speed in R.P.M.

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H = the head pumped against in feet, and

N_1 and H_1 signify the higher speed and corresponding higher head.

To see how closely the behavior of the auger pump follows that of the centrifugal pump, as set forth in the preceding law this relation may be expressed as

$$x = y^{\frac{1}{2}}$$

where $x = \frac{H}{H_1}$, and $y = \frac{N}{N_1}$

The curve on Plate VIII is the hyperbola

$x^2 = y$. From table #1 we can take the values of N and N_1 and calculate the value of $x (= \frac{N}{N_1})$; also H and H_1 and determine the values of $y (= \frac{H}{H_1})$. These values were computed and the points plotted on the curve. Since we have squared the value $\sqrt{\frac{H}{H_1}}$, as a matter of convenience in computations, and since the results when plotted lie closely to the curve $x^2 = y$, it can be seen at once that the relation $\frac{N}{N_1} = \sqrt{\frac{H}{H_1}}$ for the centrifugal pump also holds for the auger pump under the conditions of this test.

The values of N and N_1 in all cases represent the most efficient operating speed, and H and H_1 the corresponding heads pumped against. Although the maximum range of heads that the 16 inch auger pump may be operated under could not be used in these tests, due to the inadequacy of the apparatus, it may be safely concluded that the relations just proven will hold throughout the entire range of practical operating heads.

The tabulated values, from which the points were plotted are shown in Table III.

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T A B L E 111.

$$x = \frac{N}{N_1} \qquad y = \frac{H}{H_1}$$

.766	.571
.758	.607
.865	.750
.502	.260
.926	.871
.466	.226
.608	.397
.800	.654
.656	.455
.581	.346

The third object of this test was to arrive at some conclusions as to within what limits the use of the auger pump for pumping is practicable.

From observations made during the test runs the range of heads best suited for the use of the auger pump is from four to seven feet. For heads less than four feet this type of pump cannot be operated economically, although it may work satisfactorily where large discharges are desired.

The principal advantage of the auger pump for low lifts is its simplicity of design, and the ease with which it can be installed under almost any condition. Its use is particularly adapted to isolated districts and places where it will receive but little attention other than that of keeping the engine in operation, as is the case where drainage work is done. Any repairs that might be expected would quite likely be such that they could be made at any farmhouse.

A P P E N D I X A.

CALIBRATION OF THE 22.4 INCH CHANNEL WEIR.



Weir Box used in Calibrating
Channel Weir.

The channel weir was calibrated directly from the two foot weir, which had been previously calibrated by Mr. C. P. Conrad and Mr. L.H.Doolittle. Water was run from the high pressure mains into the weir box equipped with the two foot weir, for the low heads. From this weir box the water was directed into the channel of the testing apparatus discharging over the 22.4 inch channel weir. Every precaution was taken to prevent any leakages. The head of water on the two weirs was determined by simultaneous readings taken from the hook gages.

Knowing the head on the two foot weir from the calibration curve, previously drawn, the discharge in cubic feet per second could be determined. The relation between the discharge over the weir and the head on the weir crest may be expressed by a parabola of the form $x = cy^n$. If the logarithms of the head and discharge be plotted a straight line variation is shown. The logar-

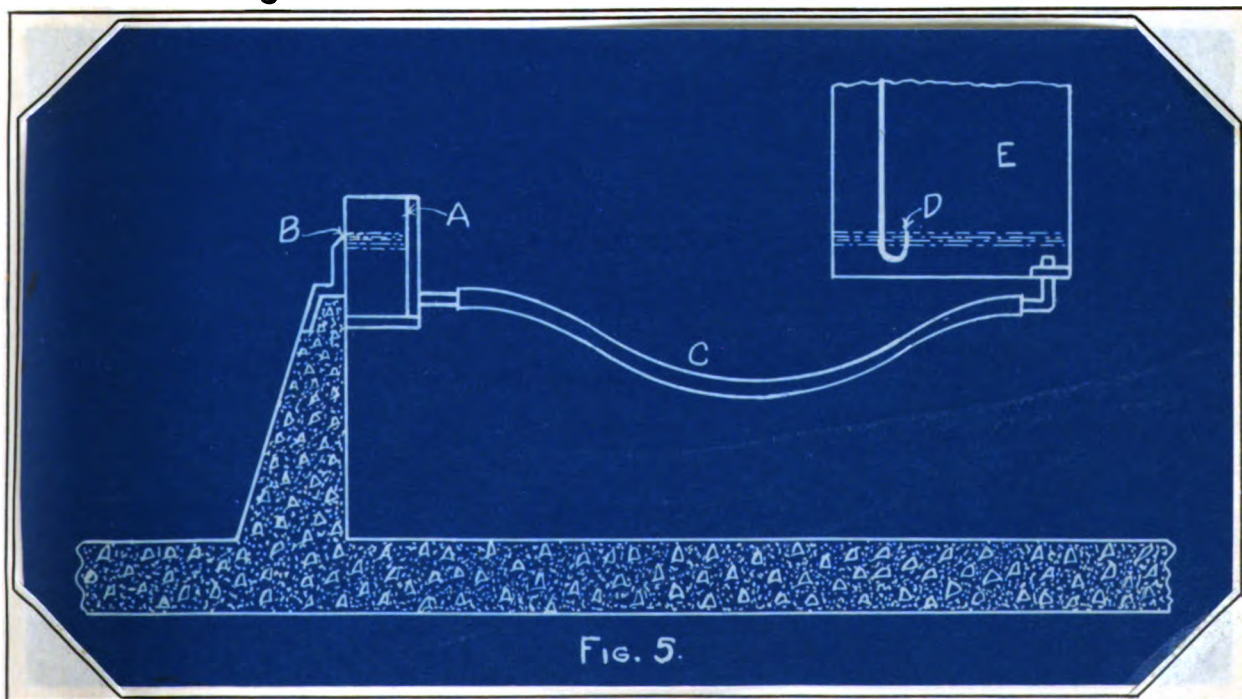
ithms of the heads on the 22.4 inch weir and the logarithms of the discharge in cubic feet per second, as obtained from the rating curve of the previously calibrated two foot weir, were plotted. From the straight line thus obtained the values for plotting the rating curve were taken by reading off the logarithms and plotting the anti-logs. The rating and logarithm curves are shown in Plate IX.

APPENDIX B.

DETERMINATION OF THE ZERO OF THE HOOK GAGE.

One method used for determining the zero of the hook gage, was to let water into the channel until it stood slightly above the weir crest. Then the chamber sluice-gate was opened a very small amount so that the water level was slowly lowered. One man was stationed over the weir and, with the aid of a light for magnification purposes, noted when the knife edge of the weir just showed at the surface of the water, at which time another man recorded the hook gage reading. This was repeated many times and an average value used in the computations.

Another method was one devised by Prof.C.I.Corp, of the Hydraulic Department. The apparatus used is shown in Fig. 5.



A small trough "A", about $1/4$ the length of the weir, was clamped so that its top was slightly above the weir crest "B", as shown in the sketch, Fig.5. A rubber tube "C" connected the trough and the hook gage stillbasin "E". Water was poured into the still basin, from whence it flowed through the rubber tubing and out over the crest of the weir until the water stood at the same level as the crest of the weir. Time was then allowed for a condition of equilibrium before the hook gage was read by raising or lowering the point "D" until it just punctured the water surface. The average reading obtained from the different methods was 0.0398 feet, which was used in the computations.

A P P E N D I X C.

DETERMINATION OF THE ZERO OF THE FLOAT GAGES.

The zero of the float gages was determined by filling the chamber with water to different heights. When the water in both the channel and the chamber was at the same level, readings were taken on both the gages. An average of the differences of the two readings, which was found to be 3.594 feet, was used in making the computations.

Plate I
 6½" Pulley
 Average R.P.M. = 302
 Runs 121-125

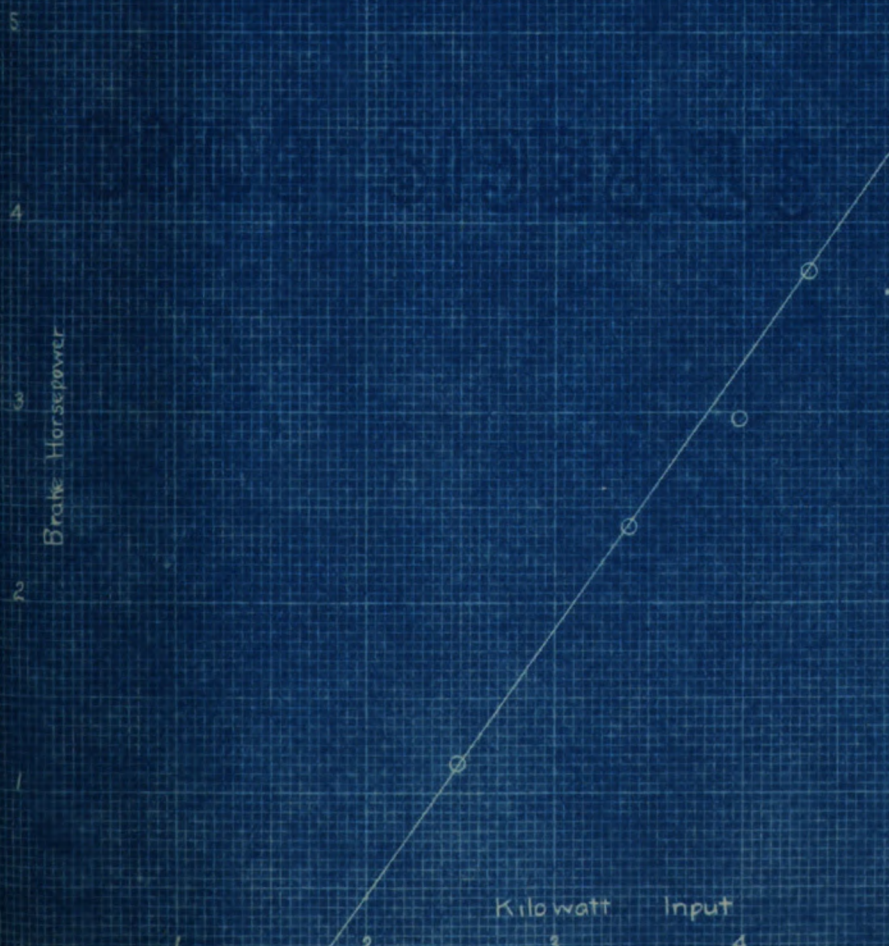


Plate II
9" Pulley
Average R.P.M. 386
Runs 1-12

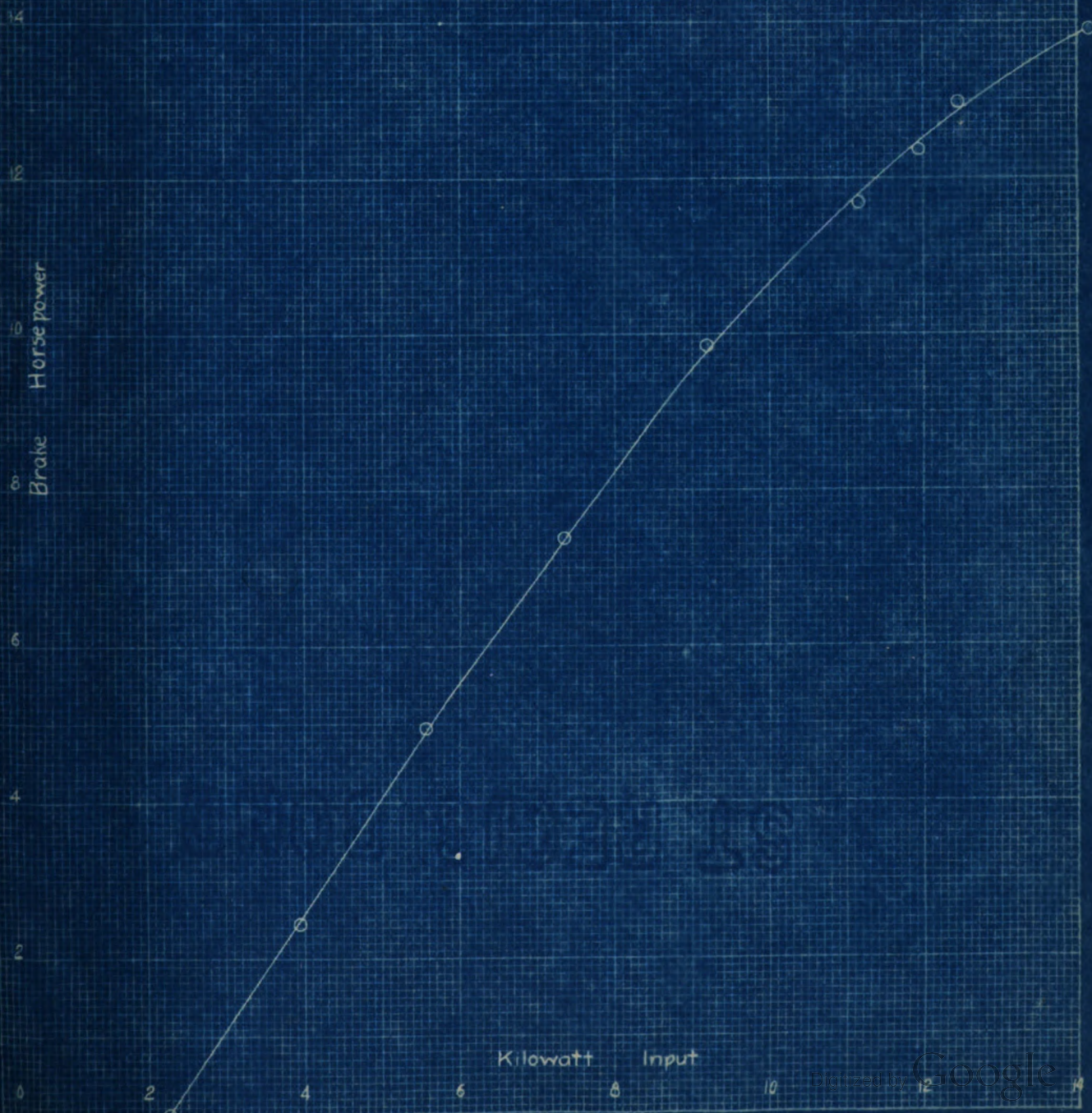


Plate III

12" Pulley

Average R.P.M. 513

Runs 27-39

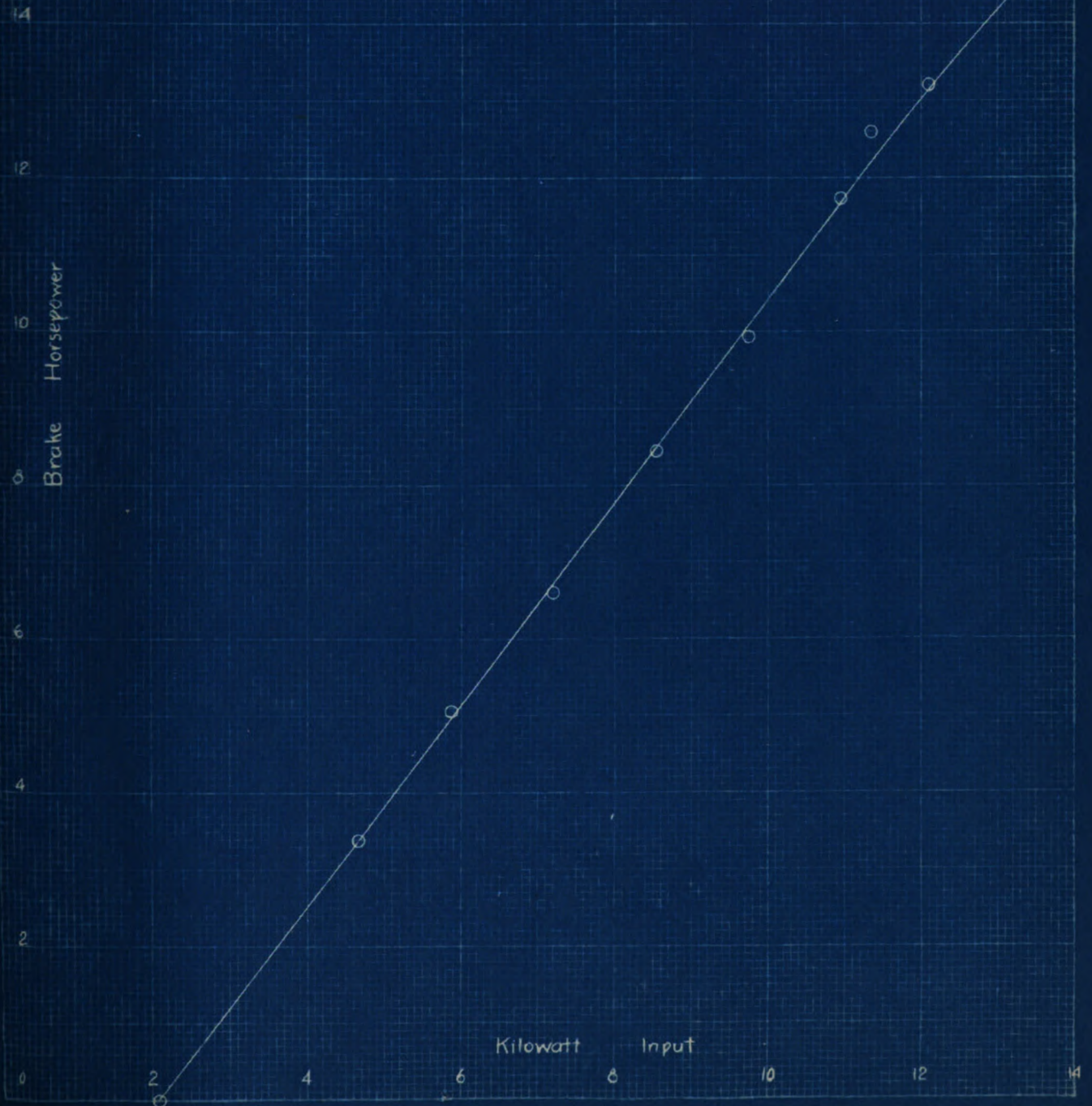


Plate IV
14" Pulley
Average R.P.M. = 604
Runs 95-101

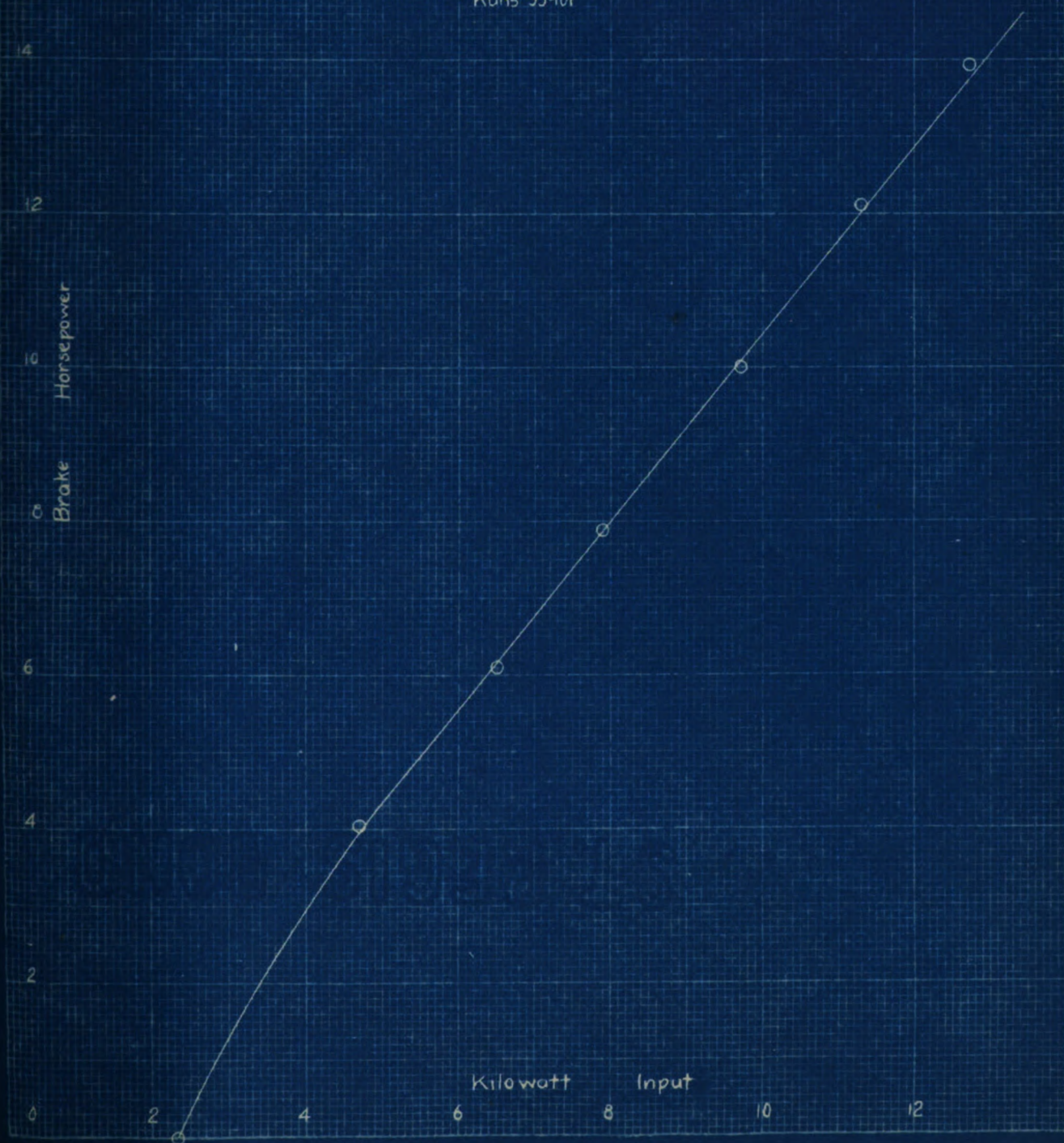


Plate V
15" Pulley
Average R.P.M. 644
Runs 40-47

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12

10

8

6

4

2

0

Brake
Horsepower

Kilowatt Input

2 4 6 8 10 12

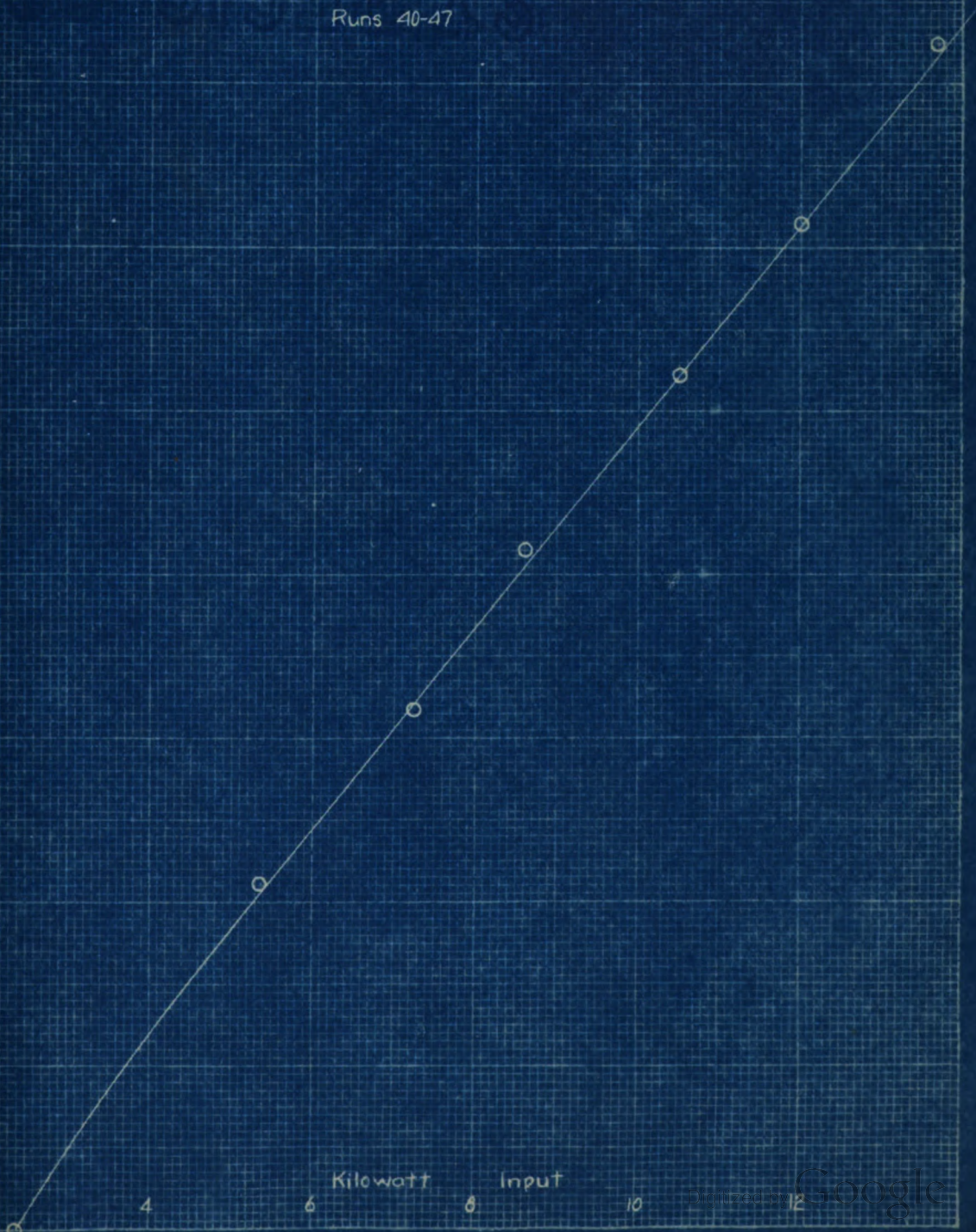
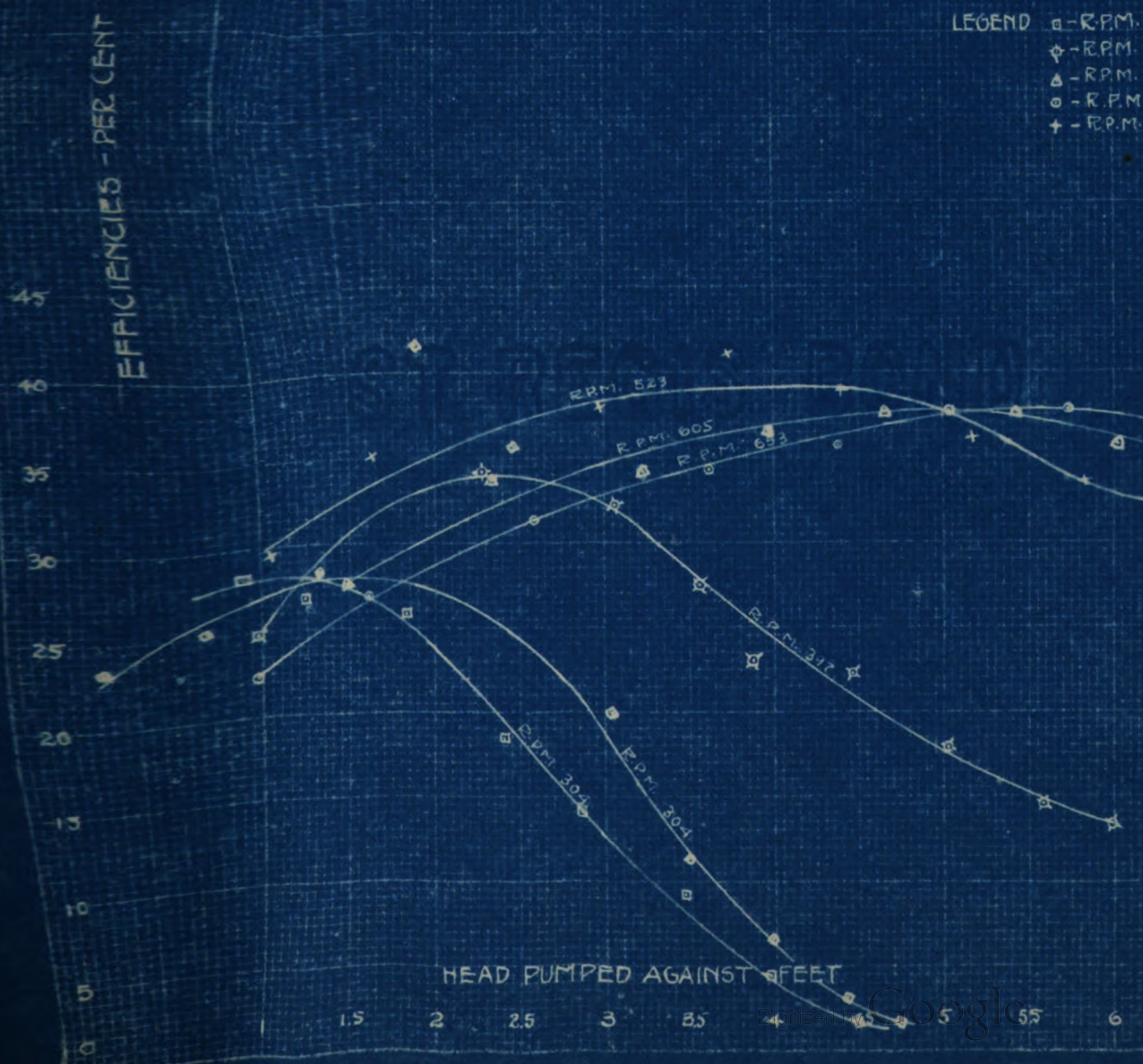
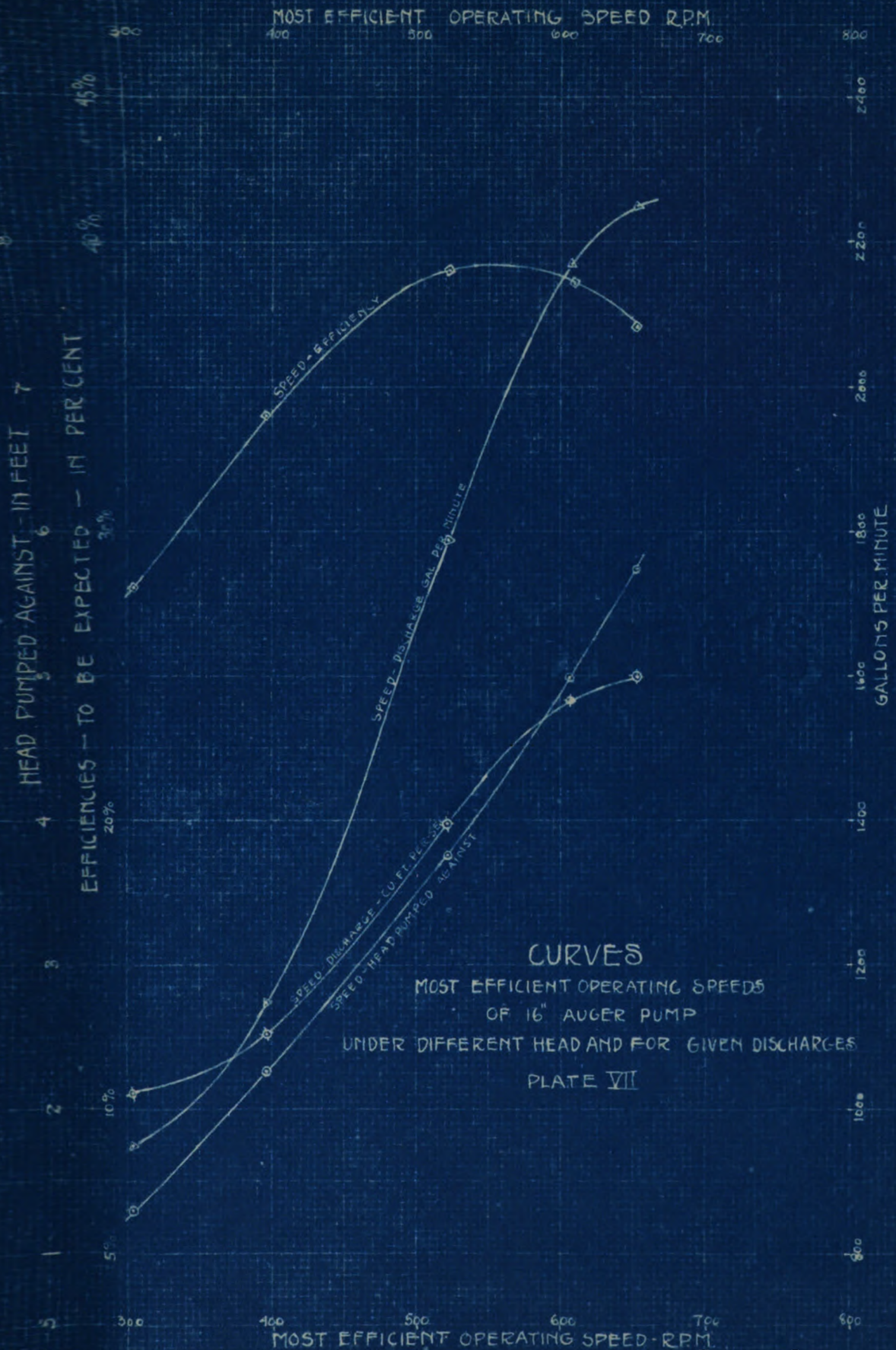


PLATE VI

CURVES

SHOWING EFFICIENCIES OF A 16" AUGER PUMP
OPERATED AT VARIOUS SPEEDS
AND AT DIFFERENT HEADS PUMPED AGAINST.



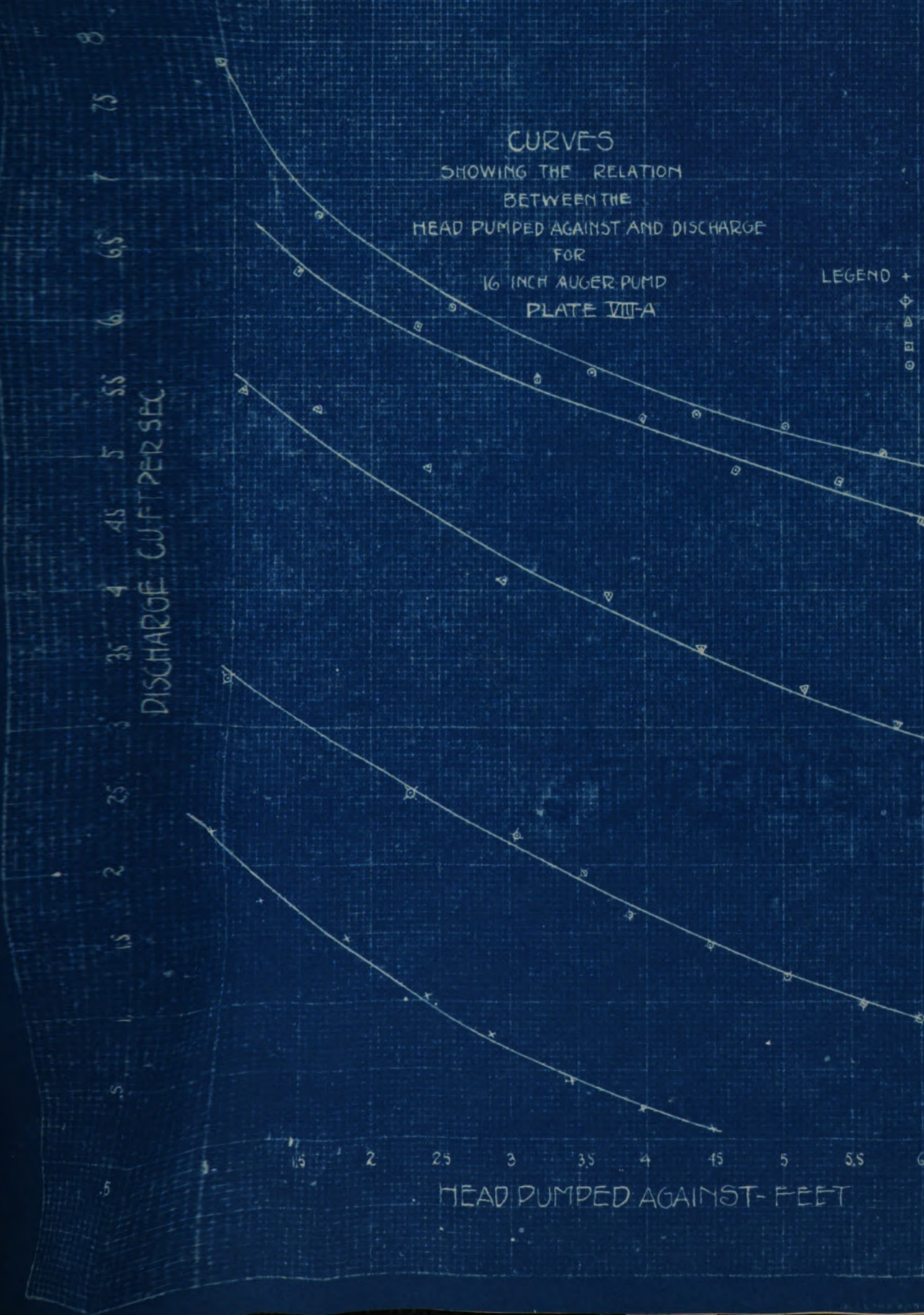


CURVES
SHOWING THE RELATION
BETWEEN THE
HEAD PUMPED AGAINST AND DISCHARGE
FOR
16 INCH AUGER PUMP
PLATE VIII-A

LEGEND +

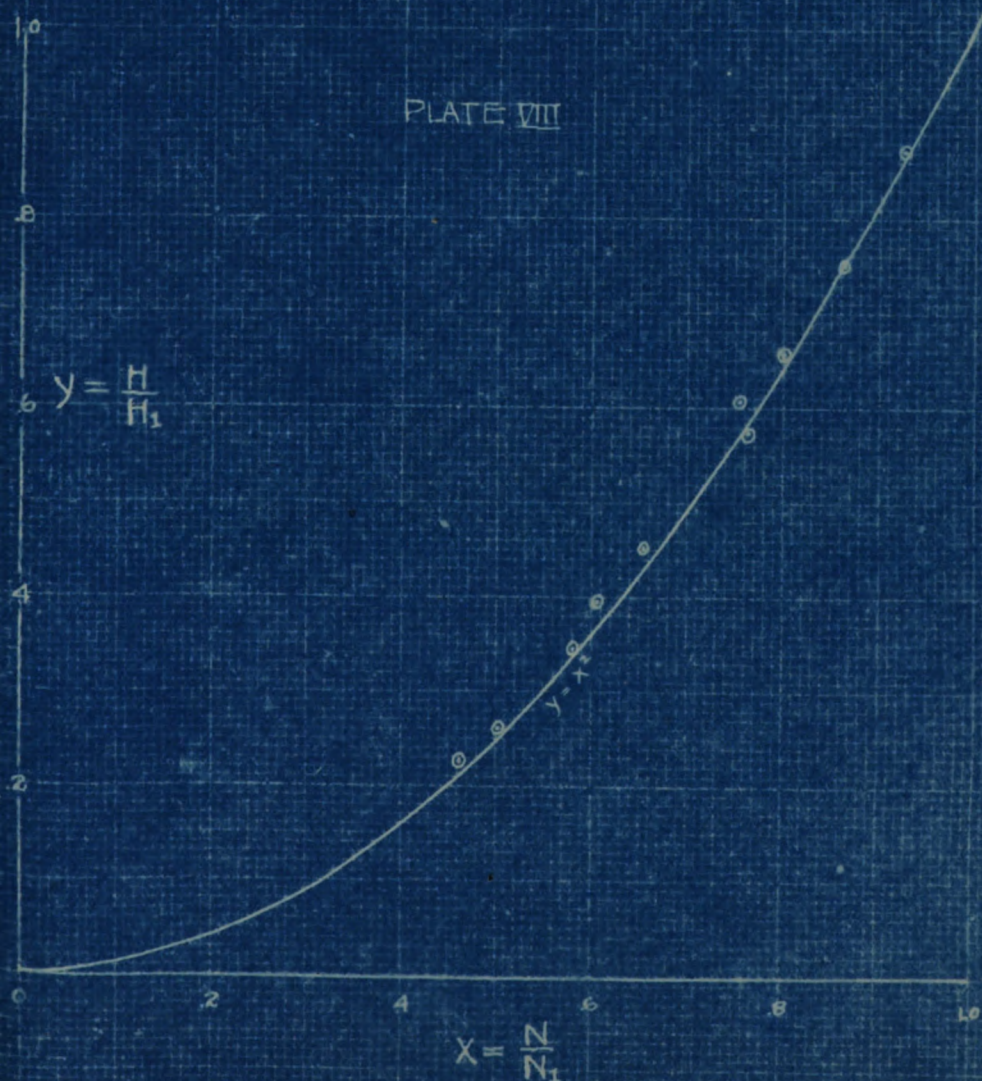
DISCHARGE CU FT PER SEC.

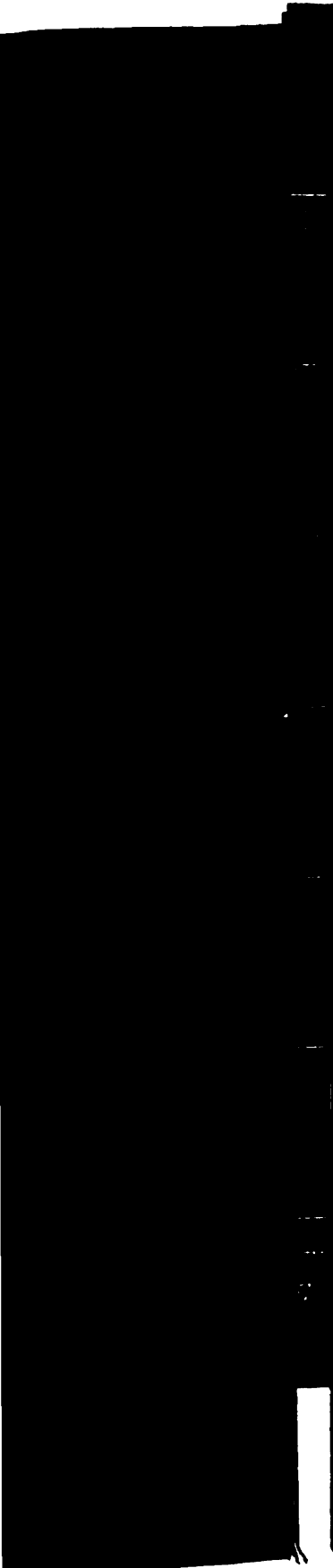
HEAD PUMPED AGAINST- FEET



ST. PETERS BO

PLATE VIII





B I B L I O G R A P H Y.

- R. L. Daugherty Hydraulic Turbines. Chap.X, p.98,
Chap. XIV, p. 135.
- I. P. Church Hydraulic Motors, Chap. VII, p. 168.
- Charles H. Innes Centrifugal Pumps; Turbines and
Water Motors, Chap. XXV - XXX.

CRANBERRY MARSH 16 INCH AUGER PUMP

WILTERDING, STIVERS, ANDERSON

1914-1915

BRAKE ARM - 41 INCHES

ZERO OF HOOK GAGE - 0.0398'

ZERO OF FLOAT GAGE - 3.594'

2	3	4	5	6	7	8	9	10	11	12	13	14
RUN NO.	AMPERES	VOLTS	KILOWATTS #3 x #4	USEFUL H.P. [FROM CURVE]	DIFF. IN HEAD PUMPED AGAINST FEET	HEAD OF WATER ON WEIR CREST	DISCHARGE, IN POUNDS PER SECOND	H.P. DELIVERED #7 x #9 550	NET BRAKE LOAD. POUNDS	B.H.P.	R.P.M.	EFFICIENCY #10 x #12
1	7.546	519.9	3.917						10	2.49	383	
2	10.95	507.9	5.560						20	4.99	383	
3	14.52	506.4	7.360						30	7.40	378	
4	18.39	501.0	9.210						40	9.85	378	
5	22.75	525.5	11.950						50	12.35	379	
6												
7	4.22	532.5	2.250		9	INCH	PULLEY		0	0	402	
8												
9	22.77	547.9	12.470						50	12.92	397	
10												
11	27.35	522.2	14.290						55	13.90	388	
12	21.15	543.7	11.150						45	11.70	399	
13	6.77	520.8	3.523	1.90	2.279	0.5125	157.0	0.650			384	34.
14	7.35	528.4	3.883	2.38	3.055	0.4711	138.0	0.766			393	32.
15	8.46	515.9	4.370	3.13	3.870	0.3853	102.5	0.720			393	23.
16	8.42	522.0	4.400	3.20	4.467	0.3482	88.1	0.715			400	22.
17	9.29	520.2	4.830	3.81	5.032	0.3093	74.3	0.680			401	17.
18	9.86	519.9	5.125	4.28	5.588	0.2702	61.2	0.622			405	14.5
19	10.09	519.5	5.230	4.45	5.987	0.2502	54.6	0.595			406	13.
20	5.95	552.4	3.290	1.48	0.971	0.6262	209.0	0.368			394	24.
21	7.71	541.9	4.180	2.85	3.550	0.4316	121.2	0.783			396	27.
22-26												
27	8.78	531.2	4.660						10	3.39	520	
28	14.27	522.4	7.460						20	6.70	515	
29	18.94	522.6	9.960						30	9.94	509.7	
30	22.03	515.1	11.360						35	11.60	508	
31	26.35	513.9	13.50						45	14.85	507	
32												

2	3	4	5	6	7	8	9	10	11	12	13	
RUN NO.	AMPERES	VOLTS	KILOWATTS #3 x #4	USEFUL H.P. [FROM CURVE]	DIFF. IN HEAD PUMPED AGAINST FEET	HEAD OF WATER ON WEIR CREST	DISCHARGE, IN POUNDS PER SECOND	H.P. DELIVERED #7 x #9 550	NET BRAKE LOAD. POUNDS	B.H.P.	R.P.M.	EFFICIENCY
33	11.30	5205	5.880			12 INCH PULLEY			15	5.06	5185	
34	16.38	5213	8.530						25	8.45	5193	
35	21.20	517.7	10.950						35	11.75	515.0	
36	4.01	524.0	2.100						0	0	5280	
37	13.60	5295	7.200						20	6.61	5093	
38	18.60	5263	9.780						30	9.91	507.0	
39	23.10	523.8	12.060						40	13.23	508.0	
40	10.05	532.7	5.350						10	4.23	650	
41	16.40	523.6	8.590						20	8.31	638	
42	23.20	518.6	12.030						30	12.28	628	
43	26.40	518.9	13.700						35	14.50	635	
44	13.80	526.0	7.250			15 INCH PULLEY			15	6.35	652	
45												
46	4.61	518.5	2.390						0	0	660	
47	20.30	519.9	10.500						25	10.45	644	
48	9.78	535.4	5.240	3.91	.978	1.1139	489.0	0.871			643	27
49	10.84	529.5	5.740	4.56	1.605	0.9977	421.0	1.227			639	20
50	12.63	527.1	6.670	5.69	2.597	0.9341	379.0	1.787			644	31
51	14.01	532.1	7.450	6.70	3.614	0.8877	350.0	2.300			657	34
52	15.09	529.9	8.000	7.38	4.384	0.8539	331.0	2.638			652	30
53	15.81	532.9	8.430	7.88	5.035	0.8429	325.0	2.975			659	31
54	16.75	536.7	9.000	8.60	5.742	0.8214	312.2	3.260			658	31
55	18.61	525.7	9.800	9.58	6.439	0.7930	297.0	3.510			664	31
56	6.92	543.3	3.760	2.22	1.061	0.8769	341.0	0.658				2
57	7.52	541.7	4.170	2.75	1.603	0.8565	332.5	0.968				3
58	8.50	540.0	4.590	3.28	2.402	0.8023	275.0	1.200				3
59	9.15	541.0	4.950	3.75	2.967	0.7670	256.0	1.380				3
60	9.70	538.3	5.220	4.12	3.746	0.7046	248.5	1.690				4
61	10.59	530.3	5.610	4.62	4.416	0.6547	223.0	1.790				3
62	11.52	530.9	6.120	5.32	5.157	0.6197	205.0	1.920				3
63	12.47	529.8	6.600	5.90	5.828	0.5846	188.0	1.990				3
64	13.04	529.6	6.900	6.33	6.205	0.5697	181.0	2.045				3
65	13.28	514.5	6.830	6.23	6.382	0.5525	173.5	2.015				3
66-78	WEIR CALIBRATION											

2	3	4	5	6	7	8	9	10	11	12	13	14
RUN NO.	AMPERES	VOLTS	KILOWATTS *3 x *4	USEFUL H.P. [FROM CURVE]	DIFF. IN HEAD PUMPED AGAINST FEET	HEAD OF WATER ON WEIR CREST	DISCHARGE, IN POUNDS PER SECOND	H.P. DELIVERED *7 x *8 550	NET BRAKE LOAD POUNDS	B.H.P.	R.P.M.	EFFICIENCY *10 x 100 = % *6
79	6.35	521.4	3.320						10	1.85	284.0	
80	9.02	526.2	4.750						20	3.69	283.3	
81	11.71	528.1	6.190						30	5.55	284.2	
82	14.57	521.2	7.600						40	7.38	283.5	
83	16.90	536.5	9.060						50	9.36	287.5	
84	19.87	538.4	10.700						60	11.33	290.5	
85	6.29	542.6	3.410						0	0	300.5	
86	4.22	542.2	2.285	0.82	0.915	0.5119	138.7	0.231			291.0	28.2
87	4.34	536.0	2.325	0.91	1.237	0.4409	108.6	0.245			298.0	26.9
88	4.69	535.4	2.515	1.17	1.833	0.3963	91.0	0.303			298.0	25.9
89	5.21	538.0	2.805	1.52	2.401	0.3235	64.4	0.281			302.0	18.5
90	5.41	537.0	2.910	1.73	2.860	0.2672	46.8	0.243			304.0	14.0
91	5.66	532.5	3.019	1.83	3.464	0.1897	26.3	0.166			305.0	9.1
92	6.00	531.0	3.186	2.12	3.966	0.1241	13.1	0.095			304.5	4.5
93	6.31	531.5	3.350	2.35	4.490	0.0712	5.2	0.042			306.0	1.8
94	6.78	529.3	3.590	2.62	5.012	—	0	0			308.0	—
95	8.95	528.2	4.730						10	4.01	602.7	
96	15.10	522.0	7.880						20	7.91	594.3	
97	21.44	527.1	11.300						30	12.12	600.7	
98	24.58	519.3	12.750						35	13.95	599.5	
99	18.82	517.9	9.750						25	10.01	603.0	
100	12.42	524.6	6.510						15	6.13	613.0	
101	4.50	524.6	2.360						0	0	624.0	
102	8.90	525.4	4.675	3.85	1.479	0.9590	395.0	1.062			603.3	27.6
103	9.95	528.3	5.260	4.66	2.332	0.9206	371.0	1.572			611.3	33.7
104	11.40	547.5	6.240	5.95	3.328	0.8802	346.0	2.030			636.5	34.1
105	12.50	535.0	6.695	6.08	3.978	0.8484	328.5	2.374			591.5	36.6
106	13.20	525.2	6.942	6.83	4.662	0.8076	304.0	2.574			613.7	37.6
107	14.40	548.5	7.812	7.85	5.432	0.7994	300.0	2.960			604	37.7
108	15.30	552.5	8.460	6.20	6.026	0.7666	281.5	3.085			638.3	35.8
109	16.20	553.4	8.960	9.20	6.496	0.7455	271.0	3.195			624.0	34.7
110	15.90	552.3	8.875	9.05	6.440	0.7519	273.5	3.200			533.0	35.3
111	4.47	556.7	2.490	1.10	1.304	0.4594	132.2	0.311			300.0	28.3

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2	3	4	5	6	7	8	9	10	11	12	13	14
RUN No.	AMPERES	VOLTS	KILOWATTS #3 x #4	USEFUL H.P. [FROM CURVE]	DIFF. IN HEAD PUMPED AGAINST FEET	HEAD OF WATER ON WEIR CREST	DISCHARGE, IN POUNDS PER SECOND	H.P. DELIVERED #7 x #9 <u>550</u>	NET BRAKE LOAD POUNDS	B.H.P.	R.P.M.	EFFICIENCY #10 x 100 = % #6
112	4.30	526.3	2.266	0.85	1.895	0.3866	103.0	0.355			300.0	41.7
113	4.32	549.9	2.375	0.99	2.468	0.3202	78.6	0.353			305.0	35.7
114	5.12	530.7	2.720	1.45	3.045	0.2446	53.0	0.294			305.0	20.3
115	5.78	532.0	3.080	1.98	3.497	0.1884	35.6	0.226			305.0	11.4
116	6.17	527.2	3.255	2.22	3.996	0.1275	20.3	0.148			306.0	6.66
117	6.32	523.0	3.310	3.30	4.421	0.0948	12.5	0.100			306.0	3.03
118	6.69	529.1	3.545	2.61	4.746	0.0468	5.0	0.043			308.0	1.65
119	4.47	535.3	2.395	1.01	0.775	0.5707	181.5	0.256			315.5	25.30
120	4.12	534.4	2.200	0.73	0.354	0.7362	264.0	0.170			314.0	23.30
121	8.355	519.3	4.340						20	3.75	2880	
122	7.604	524.8	3.990						16	2.95	283.0	
123	6.520	520.6	3.390						12	2.40	308.0	
124	4.740	525.5	2.492						6	1.16	296.3	
125	3.303	527.5	1.742						0	0	302.0	
6.5 INCH PULLEY RATING CURVE PLOTTED FROM THESE NOTES USED IN RUNS 86-94 & 111-120.												

UNIVERSITY OF WISCONSIN

COLLEGE OF ENGINEERING

DEPARTMENT OF HYDRAULIC ENGINEERING

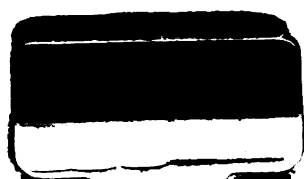
Approved

Chas. J. Corp

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